Abstract

In conventional VCHP applications either the heat dissipation of the device being maintained at constant temperature or the ambient temperature varies, however, in photonic applications both can vary. This makes VCHP based temperature control of photonics challenging. The thermal resistance of a VCHP varies with changes in both ambient temperature and heat load as the non-condensable gas expands and contracts in the condenser causing an accompanying change in the condenser active length. By actively heating the reservoir the temperature control offered by a VCHP can be enhanced. Using existing theory this paper investigates the use of wicked and non-wicked reservoirs and the effect of reservoir volume on the sensitivity of the evaporator temperature to changes in both ambient temperature and heat load for both heated and unheated reservoirs. The paper also investigates the effectiveness of the use of a steel collar between the reservoir and the condenser in reducing the heat loss to ambient. The paper concludes that passive control of evaporator temperature can be achieved for the case of a variable heat load, but not for the case of a variable ambient temperature, that evaporator temperature is much more sensitive to reservoir temperature for a wicked than a non-wicked reservoir, and that the use of a steel collar between the reservoir and the condenser significantly reduces the power required for active evaporator temperature control.

Key Word – Variable conductance heat pipes (VCHP).

Nomenclature

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>A</td>
<td>Cross-sectional area</td>
<td>m^2</td>
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<tr>
<td>A_v</td>
<td>Vapor core area</td>
<td>m^2</td>
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<tr>
<td>k</td>
<td>Thermal conductivity</td>
<td>W/(m °C)</td>
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<tr>
<td>L_a</td>
<td>Active length</td>
<td>m</td>
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<tr>
<td>L_c</td>
<td>Condenser length</td>
<td>m</td>
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<tr>
<td>L_{ia}</td>
<td>Inactive length</td>
<td>m</td>
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<tr>
<td>m_g</td>
<td>mass of non-condensable gas in reservoir</td>
<td>kg</td>
</tr>
<tr>
<td>m_{g.c}</td>
<td>mass of non-condensable gas</td>
<td>kg</td>
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<tr>
<td>P_{sat}(T)</td>
<td>Saturation pressure at temperature T</td>
<td>Pa</td>
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<tr>
<td>Q</td>
<td>Heat load</td>
<td>W</td>
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<tr>
<td>Q_{max}</td>
<td>Maximum Heat load</td>
<td>W</td>
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<td>Q_r</td>
<td>Heat lost from reservoir</td>
<td>W</td>
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<tr>
<td>R_g</td>
<td>Universal gas constant</td>
<td>J/(mol °C)</td>
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<tr>
<td>T</td>
<td>Wall temperature</td>
<td>°C</td>
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<tr>
<td>T_a</td>
<td>Active length temperature</td>
<td>°C</td>
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<tr>
<td>T_{a,set}</td>
<td>Active length setpoint temperature</td>
<td>°C</td>
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<td>T_o</td>
<td>Ambient temperature</td>
<td>°C</td>
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<tr>
<td>T_{o,max}</td>
<td>Maximum Ambient temperature</td>
<td>°C</td>
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<tr>
<td>T_r</td>
<td>Reservoir temperature</td>
<td>°C</td>
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<tr>
<td>U</td>
<td>Heat transfer coefficient</td>
<td>W/(m °C)</td>
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<tr>
<td>V_c</td>
<td>Condenser volume</td>
<td>m^3</td>
</tr>
<tr>
<td>V_g</td>
<td>Total gas volume</td>
<td>m^3</td>
</tr>
<tr>
<td>V_r</td>
<td>Reservoir Volume</td>
<td>m^3</td>
</tr>
<tr>
<td>x</td>
<td>Distance from reservoir</td>
<td>m</td>
</tr>
<tr>
<td>θ</td>
<td>Temperature difference (T-T_o)</td>
<td>°C</td>
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Subscripts

Cu: Copper
W: Wick
SS: Stainless Steel

I. Introduction

Reconfigurable optical add/drop modules (ROADMs) are optical devices that route light containing information (voice, video, and/or data) among optical fibers in telecommunications networks. ROADMs employ optical components that are temperature dependent and require local heating. Integrated on the same die as these heated components are, however, temperature sensitive components, such as optical filters, which must be maintained within narrow temperature ranges (e.g., ± 0.1 °C) to function properly [1].

The conventional solution to such problem is to mount the ROADM device on a heat spreader which, in turn, is mounted on a thermoelectric module (TEM) to control its
The objectives of this paper are:

1. To determine if it is theoretically possible to provide complete passive temperature control for fixed ambient temperature with a variable heat load and for a fixed heat load with variable ambient temperature.
2. To determine the effect of reservoir volume on a VCHP with an actively heated wicked reservoir.
3. To determine the effect of reservoir volume on a VCHP with an actively heated non-wicked reservoir.
4. To determine the ideal VCHP reservoir size and configuration for an actively heated reservoir.
5. To determine the reduction in heat loss from the reservoir by inserting a stainless steel tube between the reservoir and the condenser.

II. Heat Pipes

A constant conductance heat pipe (CCHP) is a heat transfer device with an extremely high effective thermal conductivity, i.e., 10 or more times that of Cu. It essentially consists of a hollow tube with a wick along the internal wall of the tube as illustrated in Fig. 1. The wick is then filled with a working fluid, such as water or methanol, and the ends of the CCHP are then sealed.

When one end of the CCHP is heated and the other is cooled the liquid will evaporate at the hot end (subsequently referred to as the evaporator end) and condense at the cool end (subsequently referred to as the condenser end). As the vapor condenses its latent heat of vaporization is released. The depletion of liquid from the evaporator will cause cavities to form on the internal wick surface due to the liquid clinging to the wick. As the vapor condenses on the surface of the condenser it causes the wick at the condenser to become flooded. The surface tension on the concave liquid-vapor interface in the evaporator causes the pressure of the vapor to be higher than that of the liquid while at the condenser the pressure of the vapor and the liquid are nearly equal. This creates a pressure gradient between the liquid at the condenser and the liquid at the evaporator, which drives the condensate from the condenser to the evaporator through the wick by capillary action.

III. Introduction to VCHPs

The effective conductance of a CCHP is constant regardless of variation in ambient temperature or heat load. Eq. (1) relates the heat transported by a CCHP to the evaporator temperature and the ambient temperature.

\[ Q = U L_c (T_a - T_e) \]  

As \( U \) and \( L_c \) are constant for CCHPs it is clear from Eq. (1) that for a change in ambient temperature there is an equal change in evaporator temperature whilst for a change in heat load there is a proportional change in the evaporator temperature. Using VCHPs it is possible to control the evaporator temperature by varying the effective conductance of the heat pipe in order to compensate for changes in ambient temperature and heat load.

The most common type of VCHP is the gas-loaded heat pipe. A Gas-loaded VCHP is one that has a reservoir, filled with a non-condensable gas (such as nitrogen or argon), attached to the condenser end of the heat pipe as shown in Fig. 2. By varying the volume of gas in the reservoir the
active length and thus the effective conductivity of the gas-loaded VCHP can be varied. By changing the effective conductivity of the VCHP depending on the heat load and ambient temperature it is possible to control the evaporator temperature as described in more detail below.

Fig. 2 Schematic of a Variable Conductance Heat Pipe with an Unheated Wicked Reservoir

Low ambient temperatures or low heat loads cause the active length temperature and thus the active length pressure to be low. This causes the gas in the reservoir to expand into the condenser until equilibrium is reached whereby the total pressure throughout the condenser is constant where the gas filled region is referred to as the inactive condenser section. The non-condensable gas acts as a buffer preventing the vapor flow from entering the inactive region and condensing on the wall of the inactive condenser length. As the vapor cannot condense and release its latent heat of vaporization in the inactive region there is effectively no heat transferred in the inactive region of the condenser. Thus by varying the active length of the condenser it is possible to control the effective thermal conductivity of the VCHP. This change in the conductivity allows the evaporator temperature to be controlled with changes in heat load and ambient temperature.

IV. The Flat Front Model

The VCHP configurations under consideration in this paper were modeled using the flat front model. The flat front model makes the following assumptions [6].

1. There is a flat front between the pure vapor in the active length and the non-condensable gas and vapor in the inactive length of the condenser.
2. The total pressure of the vapor in the condenser is equal to the total pressure in the reservoir.
3. Axial conduction along the wall and wick is assumed negligible so there is a step change in temperature across the vapor-gas interface and at the reservoir entrance as shown in Fig. 3.
4. It is assumed that there is pure vapor in the active length. Thus the active length temperature is the saturation temperature corresponding to the active length pressure.

Figure 3 plots the vapor temperature and pressure distribution, according to the flat front model, along the axial length of three different VCHP configurations.

V. VCHP with a Wicked Unheated Reservoir

A VCHP with passive control was modeled in order to determine if the necessary evaporator temperature control was possible without actively heating the reservoir. It is assumed that the inactive length and the reservoir are in thermal equilibrium with ambient temperature, i.e. $T_r = T_o$ as shown in Fig. 3.

The mass of non-condensable gas is selected such that when the heat load and ambient temperature are at a maximum the entire condenser length is active and all the gas is in the reservoir. This is the ideal mass of gas because it utilizes the entire condenser and provides evaporator temperature control for all ambient temperatures. $m_g$ was calculated using the ideal gas law as shown in Eq. (2).

$$m_g = \frac{P_{sat}(T_o) - P_{sat}(T_{a,max})}{R T_{a,max}} V_e$$  \hspace{1cm} (2)$$

As illustrated in Fig. 3 the partial pressure of the vapor in the reservoir is equal to the saturation pressure corresponding to ambient temperature as the reservoir is wicked and contains liquid. The Clausius-Clapeyron relation is used to relate the vapor temperature and pressure throughout this paper. The gas pressure in the reservoir is given by Eq. (3) as the total pressure in the reservoir is equal to that in the active length of the condenser as stated in the previous section.

$$P_r = P_{sat}(T_o) - P_{sat}(T_e)$$  \hspace{1cm} (3)$$
The heat removed by the VCHP is expressed in Eq. (4).

\[ Q = U L_{a} (T_{a} - T_{o}) \]  \hspace{1cm} (4)

When \( Q \) is at a maximum of 10 W and \( T_{o} \) is at a maximum of 65°C the entire condenser length is active (i.e. \( L_{a} = L_{c} \)). In order to maintain \( T_{a} \) at the set point of 75°C suitable values of \( L_{c} \) and \( U \) must be selected to satisfy Eq. (4). Letting \( L_{c} = 0.1 \) m and \( U = 10 \) W/m·K satisfies Eq. (4). By choosing a suitable fin geometry and airflow velocity this value of \( U \) is achievable as the heat sink fins mounted at the condenser provide the dominant radial thermal resistances at the condenser.

The active length can be defined as

\[ L_{a} = L_{c} - L_{wa} \]  \hspace{1cm} (5)

The total volume of gas in the VCHP is equal to the sum of the volume of gas in the reservoir and that of the gas in the inactive condenser length

\[ V_{g} = V_{r} + L_{wa} A_{w} \]  \hspace{1cm} (6)

where \( L_{wa} A_{w} \) gives the volume of gas in the condenser. Rearranging Eq. (6) in terms of \( L_{wa} \) and substituting into Eq. (5) gives an expression for the active length.

\[ L_{a} = L_{c} - \left( \frac{V_{r} - V_{g}}{A_{w}} \right) \]  \hspace{1cm} (7)

\( V_{g} \) can be determined using the ideal gas law. Substituting for \( V_{g} \) and rearranging Eq. (7) an expression for \( L_{wa} \) can be found.

\[ L_{wa} = L_{c} + \frac{V_{r} - m_{r} R_{r} T_{o}}{A_{w}} \]  \hspace{1cm} (8)

Substituting Eq. (8) into Eq. (4) an expression is formed where \( T_{a} \) can be determined as a function of \( Q \) to and \( T_{o} \) [6].

\[ Q = U (T_{a} - T_{o}) \left[ L_{c} + \frac{V_{r}}{A_{w}} - \frac{m_{r} R_{r} T_{o}}{(P_{sat}(T_{o}) - P_{sat}(T_{a})) A_{w}} \right] \]  \hspace{1cm} (9)

Consider the effect on \( T_{a} \) of a decreasing \( T_{o} \) where \( Q \) is constant in Eq. (9).

- If \( T_{o} \) decreases \((T_{a} - T_{o}) \) increases. Therefore the term in square brackets (the active length) must decrease if \( Q \) is constant.
- For the active length term to decrease \( \left( \frac{m_{r} R_{r} T_{o}}{(P_{sat}(T_{o}) - P_{sat}(T_{a})) A_{w}} \right) \) must increase.
- The decrease in \( T_{o} \) tends to decrease the above term. This causes the opposite to the desired result and actually increases the active length, however, the change in the numerator is quite modest in comparison with the change in the denominator.
- \( P_{sat}(T_{o}) \) is decreasing as it is a function of the decreasing \( T_{o} \). Thus for \( (P_{sat}(T_{o}) - P_{sat}(T_{a})) \) to decrease, as it must, \( P_{sat}(T_{o}) \) must decrease also.

Thus for \( Q \) to remain constant with a decrease in \( T_{o} \), the active length temperature must decrease also. However it must be noted that due to the decrease in the active length the change in \( T_{a} \) is much less than that of a CCHP subjected to the same conditions.

Equation (9) allows the effect of the reservoir volume on the performance of the VCHP to be observed. The performance of the VCHP was considered under two conditions; firstly, at a fixed ambient temperature of 65 °C with a heat load varying from 0 - 10 W, and secondly with a fixed heat load of 10 W with ambient temperature varying from -5 °C - 65 °C. Fig. 4 plots the evaporator temperature as a function of heat load for a number of different reservoir to condenser volume ratios \( (V_{r} / V_{c}) \).

Fig. 4 \( T_{a} \) versus \( Q \) at selected values of \( V_{r} / V_{c} \) when \( T_{o} = 65 \) °C

It can be seen from Fig. 4 that the presence of the reservoir greatly improves the control of the active length temperature. It is evident that in the (theoretical) limit of an
infinitely large reservoir the active length temperature may be maintained at its set point passively.

Fig. 5 plots the evaporator temperature as a function of ambient temperature for \( Q = 10 \) W. It is apparent that for the case of varying ambient conditions, even with an infinitely large reservoir, the active length temperature of the VCHP cannot be maintained at its set point passively. Indeed when \( T_o = -5 \, ^\circ C \), \( T_a \) is 30 \( ^\circ C \) below its set point in this limit.

In order to understand the limits of VCHP operation more clearly the case of an infinitely large reservoir was considered for both varying \( Q \) and \( T_o \).

Consider the case of varying \( Q \) and \( T_o = 65 \, ^\circ C \). Based on Eq. (4) an expression for the maximum heat load, whereby the entire condenser length is active and \( T_a \) is at the setpoint temperature, is given in Eq. (10):

\[ Q_{\text{max}} = U L_v \left( T_{a,\text{set}} - T_a \right) \]  

Dividing Eq. (9) by this expression gives [6]:

\[ \left( \frac{Q}{Q_{\text{max}}} \right) \left( T_{a,\text{set}} - T_a \right) = \frac{T_a - T_{a,\text{set}}}{T_{a,\text{set}} - T_a} \left( 1 + \frac{V_r}{V_c} - \frac{m_r R_s T_a}{(P_{sat}(T_a) - P_{sat}(T_{a,\text{set}}))V_c} \right) \]  

Where \( V_c = L_v A_v \). Substituting Eq. (2) into Eq. (11) gives

\[ \frac{Q}{Q_{\text{max}}} = \frac{T_a - T_{a,\text{set}}}{T_{a,\text{set}} - T_a} \left[ 1 + \frac{V_r}{V_c} \frac{P_{sat}(T_a) - P_{sat}(T_{a,\text{set}})}{P_{sat}(T_{a,\text{set}}) - P_{sat}(T_{a,\text{max}})} \right] \]  

Rearranging Eq. (12) in terms of \( V_c /V_r \), and considering the limit as \( V_c /V_r \to 0 \) gives

\[ \frac{Q_{\text{max}} (T_a - T_{a,\text{set}})}{Q (T_{a,\text{set}} - T_a)} \left[ 1 - \frac{P_{sat}(T_{a,\text{set}}) - P_{sat}(T_a)}{P_{sat}(T_{a,\text{set}}) - P_{sat}(T_{a,\text{max}})} \right] = 0 \]  

From Eq. (13) it can be seen that the only condition that satisfies this equation is when \( T_a = T_{a,\text{set}} \). Thus for a VCHP with an infinitely large reservoir the evaporator temperature remains constant with a change in heat load.

Similarly, for varying ambient temperature and constant heat load, consider Eq. (4) when \( T_o \) is at a maximum of 65 \( ^\circ C \) giving:

\[ Q = U L_v \left( T_{a,\text{set}} - T_{o,\text{max}} \right) \]  

Dividing Eq. (9) by this expression gives:

\[ \left( \frac{Q}{Q_{\text{max}}} \right) \left( T_{a,\text{set}} - T_{o,\text{max}} \right) = \frac{T_a - T_{o,\text{max}}}{T_{o,\text{max}} - T_a} \left[ 1 + \frac{V_r}{V_c} - \frac{m_r R_s T_a}{(P_{sat}(T_a) - P_{sat}(T_{o,\text{max}}))V_c} \right] \]  

Substituting Eq. (2) into Eq. (15) and rearranging the expression gives:

\[ \left( \frac{T_{a,\text{set}} - T_{a,\text{max}}}{(T_a - T_o)} \right) = \frac{1 + \frac{V_r}{V_c} - \frac{m_r R_s T_a}{(P_{sat}(T_a) - P_{sat}(T_{a,\text{set}}))V_c}}{\frac{T_a - T_{o,\text{max}}}{T_{o,\text{max}} - T_a} \left[ \frac{P_{sat}(T_{a,\text{set}}) - P_{sat}(T_a)}{P_{sat}(T_{a,\text{set}}) - P_{sat}(T_{a,\text{max}})} \right]} \]  

In the limit as \( V_c /V_r \to 0 \) Eq. (16) becomes:

\[ \left( \frac{T_{a,\text{set}} - T_{a,\text{max}}}{(T_a - T_o)} \right) = \frac{1}{\frac{T_a - T_{o,\text{max}}}{T_{o,\text{max}} - T_a} \left[ \frac{\frac{P_{sat}(T_{a,\text{set}}) - P_{sat}(T_a)}{P_{sat}(T_{a,\text{set}}) - P_{sat}(T_{a,\text{max}})}}{P_{sat}(T_{a,\text{set}}) - P_{sat}(T_{a,\text{max}})} \right]} \]  

If \( T_a \) decreases \( P_{sat}(T_a) \), as a function of \( T_o \) decreases also. In order for Eq. (17) to remain balanced \( P_{sat}(T_o) \) must decrease also. Thus, even for an infinite reservoir volume a change in the ambient temperature results in a change in the active length temperature.
The limits of passive control are illustrated in Fig. 6. It shows that with an infinitely large reservoir perfect passive control is theoretically possible for a VCHP with varying heat load but not for varying ambient temperature.

Thus for variable ambient temperature it is necessary to actively heat the gas reservoir in order to provide the necessary active length temperature control.

**VI. VCHP with a Heated Reservoir**

By adjusting the temperature of the gas reservoir it is possible to control the active length temperature even with variations in ambient temperature.

There were two possible VCHP designs considered: a VCHP with a wicked gas reservoir and a VCHP with a non-wicked reservoir. The effect of the wick in the reservoir is that the partial pressure of the methanol in the reservoir corresponds to the saturation pressure at the reservoir temperature while for the non-wicked reservoir the partial pressure of the methanol corresponds to the saturation pressure at ambient temperature. The significance of this is outlined in greater detail in the following sections.

A) VCHP with a Heated Wicked Reservoir

The mass of gas was selected such that when T_o = 65 °C the VCHP is fully active without requiring the addition of any heat to the reservoir. The total mass of gas is equal to the sum of the mass of gas in the reservoir and the gas in the condenser. Using this and the ideal gas law an expression for m_g is given in Eq. (18)

$$m_g = m_{g,v} + m_{g,c} = \frac{(P_{sat}(T_o) - P_{sat}(T_r))V_r}{R_g T_r} + \frac{(P_{sat}(T_r) - P_{sat}(T_o))L_{sat}A_v}{R_g T_r}$$  \hspace{1cm} (18)

Rearranging in terms of L_{ia} and substituting into Eq. (5) gives:

$$L_{ia} = L_c - \left[ \frac{R_g T_r}{(P_{sat}(T_o) - P_{sat}(T_r))A_v} \right] \times \left[ m_{g,v} \frac{(P_{sat}(T_r) - P_{sat}(T_o))}{R_g T_r} \right]$$  \hspace{1cm} (19)

Substituting Eq. (19) into Eq. (4) and rearranging gives [3]:

$$Q = U(T_o - T_c) \left[ L_c + \frac{V_r T_o P_{sat}(T_o) - P_{sat}(T_r)}{A_v T_r} \right] - \frac{w_r R_g T_r}{A_v (P_{sat}(T_o) - P_{sat}(T_r))}$$  \hspace{1cm} (20)

When the reservoir temperature is equal to the ambient temperature Eq. (20) reduces to Eq. (9).

Using Eq. (20) it was possible to determine the reservoir temperature required to maintain the active length at a constant temperature of 75 °C for a constant heat load and variable ambient temperature.

Consider the required T_r to maintain a constant T_a, where Q is constant and T_o is decreasing, in Eq. (20).  

- If T_o decreases, for a constant T_a, (T_a - T_o) increases. Therefore the term in square brackets (the active length) must decrease if Q is constant.
- The magnitude of the third term in the square brackets, \( \frac{w_r R_g T_r}{A_v (P_{sat}(T_o) - P_{sat}(T_r))} \), decreases as T_o decreases. As this is a negative term it has the opposite of the desired effect. It is not possible to control this term as it is a function of T_o.
- In order to compensate for this the second term in the square brackets, \( \frac{V_r T_o (P_{sat}(T_o) - P_{sat}(T_r))}{A_v T_r (P_{sat}(T_o) - P_{sat}(T_r))} \), must decrease by a greater amount.
- As the decrease in T_o already causes a decrease in the magnitude of the above term only a small increase in T_r, for a constant Q, is required to maintain T_a at 75 °C.
Thus, with active reservoir temperature control, it is theoretically possible to provide perfect active length temperature control.

Fig. 6 plots the reservoir temperature required to maintain the evaporator temperature at 75°C for a varying $T_o$ for different values of $V_r/V_c$. It can be seen from Fig. 6 that a VCHP with a large reservoir volume is very sensitive to changes in reservoir temperature. By reducing the reservoir volume the sensitivity of the VCHP to changes in reservoir temperature is reduced making it more practical to implement the same level of precision temperature control.

![Fig. 6 T_r required to maintain T_a at 75°C with varying T_o for a VCHP with a wicked reservoir](image)

An important point to mention about VCHPs with wicked reservoirs is that the reservoir temperature must not exceed the evaporator temperature. If this occurs the reservoir pressure will become greater than the active length pressure and the non-condensable gas will expand out of the reservoir and flood the heat pipe [3].

B) VCHP with a Heated Non-Wicked Reservoir

The partial pressure of the vapor in a non-wicked reservoir no longer equals the saturation pressure at the reservoir temperature because the reservoir no longer contains any liquid. The vapor enters the reservoir by way of diffusion from the nearest point where liquid is present, i.e., the reservoir entrance as shown in Fig. 7.

![Fig. 7 Schematic of a Variable Conductance Heat Pipe with a non-wicked heated reservoir](image)

Vapor diffusion is caused by the gas concentration differential between the inactive length and the reservoir. At equilibrium the vapor concentration in the inactive condenser length equals that in the reservoir. Therefore, the partial pressure of the vapor in the reservoir corresponds to the inactive length temperature [3].

The effect of removing the wick from the reservoir is that $P_{sat}(T_o)$ is replaced by $P_{sat}(T_a)$ in Eq. (20) giving [3]:

$$Q = U(T_o - T_a) \left[ L_c + \frac{V_r T_o}{A T_e} \left( \frac{w_r R_o}{A (P_{sat}(T_o) - P_{sat}(T_e))} \right) \right] \quad (21)$$

Consider the required $T_o$ to maintain a constant $T_a$, where $Q$ is constant and $T_o$ is decreasing, in Eq. (21).

- If $T_o$ decreases, and $T_a$ remains constant, $(T_a - T_o)$ increases. Therefore the term in square brackets (the active length) must decrease if $Q$ is constant.
- The decrease in $T_o$ causes $\frac{w_r R_o}{A (P_{sat}(T_o) - P_{sat}(T_e))}$ to decrease, which is undesirable.
- In order to provide the necessary decrease in the active length the second term within the active length term must decrease by a greater amount then the third term decreases.
- A large increase in reservoir temperature is required to decrease the second term within the active length term, as the partial pressure of the vapor is no longer a function of the reservoir temperature as it is in Eq. (20) for the case of a wicked reservoir.

Thus, to achieve theoretically perfect active length temperature control of a VCHP with a non-wicked reservoir a large variation in reservoir temperature is required.

Fig. 8 plots the reservoir temperature required in order to maintain the active length temperature at 75°C for a constant heat load and variable ambient temperature.

![Fig. 8 T_r required to maintain T_a at 75°C for varying T_o.](image)

It is clear from Fig. 8 that the reservoir temperatures required are not feasible even with a large reservoir. Smaller reservoir volumes would result in an even greater reservoir temperature range. This is due to Eq. (21) being dominated by the uncontrolled change in gas pressure in the third term of the inactive length term. Unlike the wicked reservoir there is no compensating increase in the second term without extreme excursions in reservoir temperature.
C) Comparison of Heated Wicked and Non-Wicked Reservoir

It is clear from Fig. 7 and Fig. 8 that the active length temperature of a VCHP with a non-wicked reservoir is very insensitive to changes in the reservoir temperature while the opposite is the case for the VCHP with a wicked reservoir.

The reason for this difference in sensitivity between the wicked and non-wicked reservoirs is that for a wicked reservoir the partial pressure of methanol equals the saturation pressure at the reservoir temperature while in the non-wicked reservoir the partial pressure of methanol equals the saturation pressure at ambient temperature. The effect of this is that it is possible to control the partial pressure of methanol vapor for a wicked reservoir while it is not possible to control the partial pressure of methanol vapor for a non-wicked reservoir. This is critical because the partial pressure of the methanol in the reservoir dominates the temperature change. This is apparent in Fig. 8 where an unrealistically large change in temperature is required to overcome the change in the partial pressure of the methanol.

VII. Heat loss from an actively heat reservoir

As per Fig. 6 the maximum reservoir temperature is when the ambient temperature is at a minimum. This results in a large temperature gradient along the inactive condenser length. It is necessary to determine this temperature gradient for two reasons.
1. To determine the heat loss from the reservoir.
2. The temperature gradient reduces the accuracy of the flat-front model and alters the heat pipe performance.

A one dimensional fin analysis was performed to determine the temperature profile along the inactive condenser length in the following sections.

A) Heat loss from an actively heat reservoir with no SS insert

The base of the fin is taken to be the point where the heat pipe is meets the gas reservoir and the tip of the fin is taken to be the pure vapor/ pure vapour + gas interface in the condenser as shown in Fig. 9 where the light grey indicates the active condenser length and the dark grey indicates the inactive condenser length and the reservoir.

The base temperature of the fin is taken to be equal to that of the reservoir while that of the tip is assumed to be at the active length temperature. The reservoir is insulated and, thus, assumed to be adiabatic. The only possible heat loss from the reservoir occurs by axial conduction along the wick and the copper wall of the condenser whereby the heat is, in turn, lost to ambient by convection.

The heat loss from the reservoir is at a maximum when ambient temperature is at a minimum of -5 °C. For this ambient temperature a reservoir temperature of 71.7 °C is required to give the necessary active length of 87.5 mm for a VCHP with a wicked reservoir. The temperature at the vapour-gas interface is assumed to be that of the active length, which is at 75 °C. Inserting these boundary conditions into Eq. (22) allows the temperature distribution of the inactive length to be plotted [7].

\[
\theta = \frac{\theta_0}{\theta_0 - \frac{\sinh mx + \sinh m(L_w - x)}{\sinh mL_w}}
\]

where

\[
m = \sqrt{\frac{U}{(kA_c + kA_w) \left(\frac{U}{T_r - T_a}\right)}}
\]

The theoretical analysis was verified by generating a two-dimensional axisymmetric FEA model of the inactive condenser length, with identical boundary conditions, using Pro/Mechanica. It is clear from Fig. 11 and Fig. 12 that there is a very close correlation between the FEA and analytical results.

Fig. 11 shows the temperature distribution calculated theoretically numerically. It also includes the temperature profile used by the flat front model.

It is apparent from Fig. 11 that the temperature distribution predicted by the FEA and the temperature
distribution calculated theoretically do not compare well with the flat front model. The large deviation of the actual temperature profile from the flat front model indicates the limitations of the flat front model.

Using Eq. (24) it is possible to determine the heat loss from the reservoir to the inactive length of the condenser [7].

\[
Q_r = M \left( \frac{\cosh ml - \left( \frac{\theta_i}{\theta_r} \right)}{\sinh ml \theta_r} \right)
\]

(24)

where

\[
M = \theta_i \sqrt{U \left( (kA)_{cs} + (kA)_{ss} \right)}
\]

(25)

From Eq. (24) it was found that the heat lost by axial conduction from the reservoir, \(Q_r\), is 17.7 W.

B) Temperature distribution along the inactive length of the condenser with a SS insert

This section determines the effect of the stainless steel insert on the inactive condenser length temperature profile and the resultant reduction in heat loss from the reservoir. For a fin of constant cross-sectional area [7]:

\[
d^2 \theta \over dx^2 - m^2 \theta = 0
\]

(26)

\(\theta\) is referred to as \(\theta_1\) along the inactive condenser length and as \(\theta_2\) along the stainless steel insert as shown in Fig. 10.

\[
\theta_1 = \theta_2 = \theta_j = T_j - T_o
\]

At the interface between the stainless steel insert and the inactive length the temperature from each solution must match. \(\theta_j\) is the difference between the wall temperature at the junction and ambient temperature giving:

\[
\theta_1 = \theta_2 = \theta_j = T_j - T_o
\]

Consider the element at the interface between the inactive length and the stainless steel insert as shown in Fig. 10. From Fourier’s law it is known that

\[
Q_m = \left( k_{ss} A_s + k_w A_w \right) \frac{d\theta_j}{dx}
\]

(30)

and that

\[
Q_{ss} = \left( k_{cs} A_{cs} + k_w A_w \right) \frac{d\theta_j}{dx}
\]

(31)

\[
\theta_i = C_i e^{-x} + C_j e^{-mx}
\]

(27)

Because in practice the stainless steel insert will be insulated it was assumed to be adiabatic. As there is no heat loss to ambient \(m = 0\) and Eq. (26) reduces to

\[
d^2 \theta \over dx^2 = 0
\]

(28)

It can be easily verified that Eq. (29) is a solution to this equation

\[
\theta_2 = D_1 x + D_2
\]

(29)

The boundary conditions are shown above in Fig. 10 and are as follows:

- The wall temperature at the vapour-gas interface was at the active length temperature and the wall temperature at the reservoir was at the reservoir temperature.

\[
@ x = L \quad \theta_1 = \theta_a = T_r - T_o
\]

\[
@ x = R \quad \theta_2 = \theta_r = T_r - T_o
\]

- At the interface between the stainless steel insert and the inactive length the temperature from each solution must match. \(\theta_j\) is the difference between the wall temperature at the junction and ambient temperature giving:

\[
@ x=0 \quad \theta_1 = \theta_2 = \theta_j = T_j - T_o
\]
From the conservation of energy it is known that \( Q_{in} = Q_{out} \). Equating Eq. (30) and Eq. (31) gives:

\[
\frac{d\theta_y}{dx} = n \frac{d\theta_i}{dx} \tag{32}
\]

where

\[
n = \frac{k_{c \alpha} A_{c \alpha} + k_{w \alpha} A_{w \alpha}}{k_{SS} A_{SS} + k_{w} A_{w}}
\]

By applying these boundary conditions and manipulating the resulting equations it is possible to determine the value of the constant coefficients in Eq. (27) and Eq. (29) and subsequently plot the temperature profile along the stainless steel insert and the inactive condenser length as a function of \( x \) as shown in Fig. 12.

![Fig. 12 Temperature Distribution along the inactive condenser length for a VCHP with a SS insert.](image)

The temperature profile of the stainless steel insert is the region along the X-axis from \(-0.025\) to \(0\) in Fig. 13, while the remainder of the plot is the temperature distribution along the inactive length.

From Fig. 12 it is possible to determine the heat lost from the reservoir. As the stainless steel section is adiabatic and acts as an insulator to the reservoir the axial conduction is greatly reduced. From the analytical model it was calculated that the heat lost was approximately \(4.5\) W, this is greatly reduced from the heat lost without the stainless steel insert.

VIII. Conclusions

- While complete passive control is theoretically possible for fixed ambient temperature with a variable heat load, it is not possible to provide complete passive control for variable ambient temperature thus it is necessary to control the reservoir temperature.
- The evaporator temperature of VCHPs with large actively heated wicked reservoirs are very sensitive to small changes in reservoir temperature but by reducing the reservoir size it is possible to provide the necessary temperature control with a temperature range that is more practical to implement.
- A VCHP with a non-wicked reservoir requires too large a reservoir temperature range to control the evaporator temperature over the full range of ambient temperatures even with a very large reservoir.
- A VCHP with a small wicked reservoir provides the best control of the active length temperature as the reservoir temperature range is the most practical to implement.
- By inserting a stainless steel thin walled tube between the reservoir and the condenser it is possible to greatly reduce the heat loss through axial conduction along the wall of the condenser.

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References


